

Thermodynamic model to study a solar collector for its application to Stirling engines



Amir Abdollahpour^a, Mohammad H. Ahmadi^{b,*}, Amir H. Mohammadi^{c,d,*}

^a Faculty of Mechanical Engineering, K.N. Toosi University, Tehran, Iran

^b Department of Renewable Energies, Faculty of New Science and Technologies, University of Tehran, Tehran, Iran

^c Institut de Recherche en Génie Chimique et Pétrolier (IRGCP), Paris Cedex, France

^d Thermodynamics Research Unit, School of Chemical Engineering, University of KwaZulu-Natal, Howard College Campus, King George V Avenue, Durban 4041, South Africa

ARTICLE INFO

Article history:

Received 22 August 2013

Accepted 16 December 2013

Available online 31 January 2014

Keywords:

Solar collector

Optical analysis

Overall loss coefficient

Fluid temperature

Absorber tube

Heat loss

Thermal analysis

ABSTRACT

Energy production through clean and green sources has been paid attention over the last decades owing to high energy consumption and environmental emission. Solar energy is one of the most useful energy sources. Due to high investment cost of centralized generation of electricity and considerable loss in the network, it is necessary to look forward to decentralized electricity generation technologies. Stirling engines have high efficiency and are able to be coupled with solar energy which cannot be applied in internal combustion engines. Solar Stirling engines can be commercialized and used to generate decentralized electricity in small to medium levels. One of the most important steps to set up an efficient solar Stirling engine is choosing and designing the collector. In this study, a solar parabolic collector with 3500 W of power for its application to Stirling engines was designed and analyzed (It is the thermal inlet power for a Stirling engine). We studied the parabolic collector based on optical and thermal analysis. In this case, solar energy is focused by a concentrating mirror and transferred to a pipe containing fluid. MATLAB software was used for obtaining the parameters of the collector, with respect to the geographic, temporal, and environmental conditions, fluid inlet temperature and some other considerations. After obtaining the results of the design, we studied the effects of changing some conditions and parameters such as annular space pressure, type of the gas, wind velocity, environment temperature and absorber pipe coating.

© 2013 Elsevier Ltd. All rights reserved.

1. Introduction

As a non-emission and lasting energy, solar energy and its related applications has been widely studied such as solar buildings [1], solar water heating systems and solar energy conversion systems [2–19]. In solar energy systems, thermal energy storage (TES) unit is an important component to ensure the system stability and steady operation with high efficiency performance; due to dependence of its performance on weather conditions and time. Latent heat thermal energy storage (LHTES) has more density and grater temperature of constant phase change, in comparison with TES. Recently, parabolic trough power plant has become as a unique and progressed mean for solar electricity production. A parabolic trough solar collector (PTC) utilizes the energy of sun

radiation and employs its useful thermal energy for the heat transfer fluid (HTF) which circulates in the system.

By determining the geometry and thermal properties of a solar system, the thermal performance and energy obtained by the HTF can be estimated under different conditions. PTCs usually operate at 400 °C. Synthetic oil is utilized as HTF generally. Primary design of the solar power plant, prediction of thermal losses, study of collector degradation effects and HTF flow rate control require heat transfer analysis [20]. Researchers have been working on numerous heat transfer models of PTCs, since 1970s.

Edenburn [21,22] evaluated the efficiency of a PTC by using an analytical heat transfer method and the results were satisfying. Ratzel et al. [23] performed both analytical and numerical study of the heat losses in an annular receiver. Clark [24] has studied the thermal and economical performance of parabolic trough receivers. Dudley et al. [25] presented a one dimensional steady state thermal model of SEGS LS-2 parabolic collector. This model was verified with experimental data of Sandia National Laboratories (SNL) for various receiver annulus settings.

Thomas [26] has investigated a set of numerical equations based on heat transfer and heat loss analysis in a PTC receiver. A

* Corresponding authors. Addresses: Department of Renewable Energies, Faculty of New Science and Technologies, University of Tehran, Tehran, Iran (M.H. Ahmadi), Institut de Recherche en Génie Chimique et Pétrolier (IRGCP), Paris Cedex, France (A.H. Mohammadi).

E-mail addresses: mohammadhosein.ahmadi@gmail.com (M.H. Ahmadi), a.h.m@irgcp.fr (A.H. Mohammadi).

Nomenclature

A_f	the ratio of loss area to total area of concentrator	W_a	the collector aperture (m)
C_p	specific heat capacity ($\text{J kg}^{-1} \text{ } ^\circ\text{C}^{-1}$)	Greek letters	
D	diameter (m)	ν	kinematic viscosity ($\text{m}^2 \text{ s}^{-1}$)
F_R	the ratio of heat transfer to maximum heat transfer	β	volumetric thermal expansion coefficient (ideal gas) (K^{-1})
F'	collector efficiency factor	σ	Stefan–Boltzmann constant
h	convection heat transfer coefficient ($\text{W m}^{-2} \text{ } ^\circ\text{C}^{-1}$)	φ_r	rim angle
h_f	HTF convection heat transfer coefficient at T_f ($\text{W m}^{-2} \text{ } ^\circ\text{C}^{-1}$)	φ	the angle between collector axis and reflector beam
I_b	beam radiation incident on the receiver surface	ρ	surface reflectance
k	thermal conductivity ($\text{W m}^{-1} \text{ } ^\circ\text{C}^{-1}$)	ε	emissivity
g	gravitational constant (9.81 m s^{-2})	α	absorbance
$\dot{m}$	mass flow rate (kg s^{-1})	δ_s	inclination angle
Nu	Nusselt number	Acronym	
Pr	Prandtl number	SNL	Sandia National Laboratories
Ra	Rayleigh number	TES	thermal energy storage
T	temperature ($^\circ\text{C}$)	LHTES	latent heat thermal energy storage
T_p	is the mean temperature of absorber tube	PTC	parabolic trough solar collector
N	number of day from 1st January	HTF	heat transfer fluid
h_s	the hour angle		
La	latitude		
U_L	achieved through heat transfer modeling between external surface of absorber tube and environment		

one dimensional energy balanced model was designed by Forristall to evaluate heat transfer of solar receiver [27]. Stuetzle [28] presented an unsteady state analysis of solar collector by partial differential equations to determine the outlet temperature. Valladares and Velasquez [29] investigated a numerical model for a single pass solar receiver. They showed that the proposed configuration increases the thermal efficiency of the solar collector. Recently, three dimensional heat transfer analysis of PTCs was carried out by simultaneously use of the Monte Carlo Ray Trace Method (MCRT) and CFD analysis [11,30,31].

In the present work, initially, we review the solar radiation energy and the effective parameters on absorbed solar flux. These parameters depend on season, time of day, location and incident beam. Then, we analyze based on optical. Absorbed reflective flux by absorber pipe is one of the results of the optical analysis. In the next step, thermal analysis with respect to the energy equilibrium model is carried out. This model includes all the necessary equations to determine the components in the energy balance such as type of the collector, terms of absorber pipe, the optical properties and the environmental conditions. The absorbed heat is results of this analysis that are gained using MATLAB software. According to the analysis and the considered power of collector, we wrote a code in MATLAB software and we studied and designed the collector. This design is considered the geographic, temporal, and environmental conditions such as the atmospheric pressure, the environmental temperature, and wind velocity. Also, other factors like type of the fluid, the annular space pressure (between the absorber pipe and the glass coating) and some design considerations for the components of the collector including concentrator, absorber tube and glass coating on the absorber are studied and designed.

2. Optical analysis

Kalogirou studied parabolic collectors through optical analysis. The angle between incident beam on the edge of concentrating surface (where mirror radius r_r is maximum) and center line of the collector is φ_r which is called rim angle [32].

Diameter is found from following equation.

$$D = 2r_r \sin(\theta_m) \quad (1)$$

θ_m which is shown in Fig. 1, is a function of the tracking mechanism precision and reflective surface disorders [32].

$$r = \frac{2f}{1 + \cos(\varphi)} \quad (2)$$

where φ is the angle between the collector axis and the reflector beam.

When φ changes from 0 to φ_r consequently, r changes from f to r_r and D increases from $2f \sin(\theta_m)$ to $\frac{2r_r \sin(\theta_m)}{\cos(\varphi_r + \theta_m)}$. Eq. (2) in φ_r angle;

$$r_r = \frac{2f}{1 + \cos(\varphi_r)} \quad (3)$$

W_a is found from Eq. (4)

$$W_a = 2r_r \sin(\varphi_r) \quad (4)$$

where W_a is the collector aperture (m).

Through Eqs. (3) and (4)

$$W_a = \frac{4f \sin(\varphi_r)}{1 + \cos(\varphi_r)} \quad (5)$$

$$W_a = 4f \tan\left(\frac{\varphi_r}{2}\right) \quad (6)$$

In parabolic collectors, a number of reflective beams to the concentrating do not have incident to receiver, which is known as end effect and the area is illustrated in Fig. 2.

$$A_e = fW_a \tan(\theta) \left[1 + \frac{W_a^2}{48f^2} \right] \quad (7)$$

These kinds of collectors are often restricted by opaque plates which causes shadow on a part of reflector [32,38,39].

$$A_b = \frac{2}{3} W_a h_p \tan(\theta) \quad (8)$$

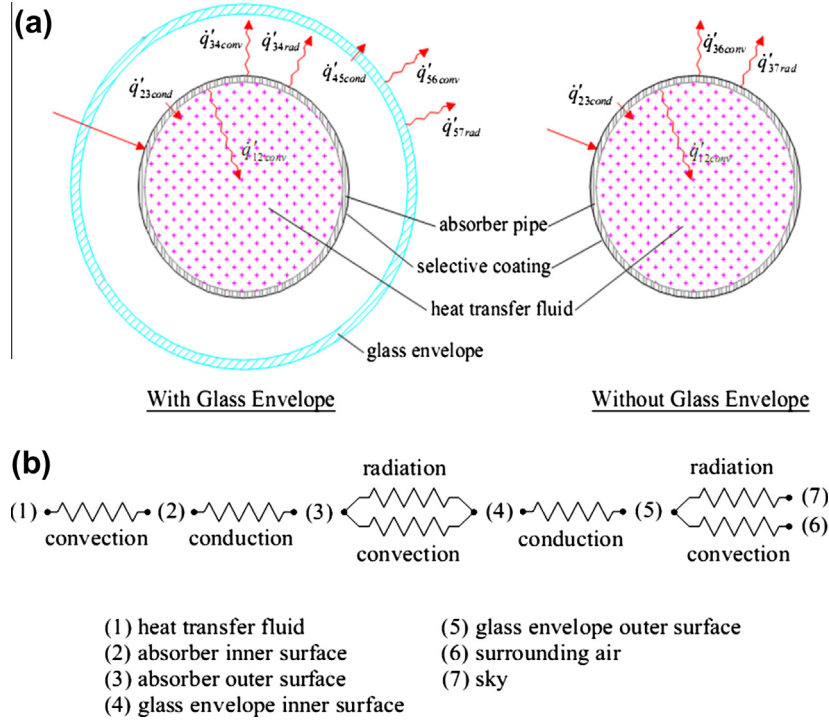


Fig. 3. (a and b) Steady state one-dimensional energy balance. Thermal resistance model [27].

3.3. Conduction heat transfer through absorber tube

Fourier's law of conduction for a hollow cylinder is shown with the following equation [34]

$$\dot{q}_{23cond} = 2\pi k_{23}(T_2 - T_3) / \ln(D_3/D_2) \quad (23)$$

The conduction coefficient is a function of the tube material. The current case comprises of three types of stainless steel, 304L, 316L and 321H and a type of Copper B42. Conduction coefficient for 304L and 316L is [35]

$$k_{23} = (0.013)T_{23} + 15.2 \quad (24)$$

for 321H

$$k_{23} = (0.0153)T_{23} + 14.775 \quad (25)$$

and for B42 is 400 (W/mK).

3.4. Heat transfer between absorber and glass envelope

The heat transfer between the absorber and the glass envelope is made up of radiation and convection. Convection heat transfer is a function of pressure [33]. When pressure is less than 1 mmHg, heat transfer is through molecular conduction unless, it is through free convection.

3.4.1. Convection heat transfer (vacuum)

In vacuum state between the absorber and the glass envelope (pressure low than 1 mmHg), convection heat transfer is negligible [33].

$$\dot{q}_{34conv} = 0$$

3.5. Convection heat transfer (with pressure)

When pressure is over 1 mmHg, the convection heat transfer is gained with the following equations [27,32].

$$\dot{q}_{34conv} = \frac{2.425k_{34}(T_3 - T_4)(PrRa_{D3}/(0.861 + Pr_{34}))^{1/4}}{(1 + (D_3/D_4)^{3/5})^{5/4}} \quad (26)$$

$$Ra_{D3} = \frac{g\beta(T_3 - T_4)D_3^3}{\alpha\nu} \quad (27)$$

$$\beta = \frac{1}{T_{avg}} \quad (28)$$

3.6. Radiation heat transfer

The radiation heat transfer between the absorber and the glass envelope is represented in the following equation [27,34]:

$$\dot{q}_{34rad} = \frac{\sigma\pi D_3(T_3^4 - T_4^4)}{(1/\epsilon_3 + (1 - \epsilon_4)D_3/(\epsilon_4 D_4))} \quad (29)$$

3.7. Conduction heat transfer through the glass envelope

For obtaining the conduction heat transfer through the glass envelope, the same equation (Eq. (23)) of the absorber are applied. It should be noticed that the temperature distribution is linear and the glass thermal coefficient is 1.04 [36].

3.8. Heat transfer between the glass envelope and environment

Heat transfer between the glass envelope and environment includes convection and radiation. The convection can be forced or natural depends on the wind condition.

3.8.1. Convection heat transfer

Eq. (30) shows the convection heat transfer between the glass envelope and atmosphere [27]

$$\dot{q}_{56conv} = h_{56}D_5\pi(T_5 - T_6) \quad (30)$$

$$h_{56} = Nu_{D_5} \frac{k_{56}}{D_5} \quad (31)$$

Here, the Nusselt number is a function of wind condition.

Case 1; without wind:

Natural convection is observed when wind is not blowing. In this case, when $10^5 < Ra_{D_5} < 10^{12}$, the following equations can be employed.

$$Nu_{D_5} = \left\{ 0.6 + \frac{0.387 Ra_{D_5}^{1/6}}{[1 + (0.559/Pr_{56})^{9/16}]^{8/27}} \right\}^2 \quad (32)$$

$$Ra_{D_5} = \frac{g\beta(T_5 - T_6)D_5^3}{\alpha_{56}\nu_{56}} \quad (33)$$

$$\beta = \frac{1}{T_{56}} \quad (34)$$

$$Pr_{56} = \frac{\nu_{56}}{\alpha_{56}} \quad (35)$$

Case 2; with wind

Forced convection is noticed when wind is blowing. In this case, when $0.7 < Pr_6 < 500$ and $1 < Re_{D_5} < 10^6$, the following equations are available (see Table 1).

$$Nu_{D_5} = C Nu_{D_5}^m Pr_6^n \left(\frac{Pr_6}{Pr_5} \right)^{1/4} \quad (36)$$

$$Pr \leq 10 : n = 0.37$$

$$Pr > 10 : n = 0.36$$

All the fluid properties are taken into account at the atmospheric temperature, except Pr_s which is at the temperature of the external surface of the glass envelope.

3.8.2. Radiation heat transfer

It is assumed that the cover is a small convex gray object in a large black body cavity, the sky. In this case, net radiation transfer between the glass envelope and sky is given by [27,37]:

$$\dot{q}_{57rad} = \sigma \pi D_5 \varepsilon_5 (T_5^4 - T_7^4) \quad (37)$$

To simplify the model, the effective sky temperature is approximated as 8 °C below ambient temperature.

4. Result and discussion

4.1. Thermal loss calculation

The prominent point in thermal loss is that thermal loss from external surface of the absorber tube to the glass envelope has to be equal to the thermal loss from the glass envelope to the environment. In this regard, with initial fluid temperature and initial guess for U_L , Eqs. (11), (12), (12a), (13)–(37) are

solved and the mean temperature of the absorber tube is found. To find the glass envelope temperature, we used an iterative technique. First, we assumed a value for U_L (initial guess for U_L is 1 in our code) and then assumed a value for glass envelope temperature. An initial guess near the environmental temperature must be made for the glass envelope temperature and it was 30 °C as an initial guess. If the thermal difference between the external surface of the absorber tube and the glass is not equal to the thermal difference between the glass surface and the environment, the procedure needs to go onto reach an equal thermal loss. Afterward, if the initial U_L is different with U_L through following equation, it has to be started from the first point. It is clear, for any assumed value of the U_L , a value for the mean temperature of absorber pipe is obtained by considering Eq. (18).

$$U_L = \frac{(\dot{q}_{34conv} + \dot{q}_{34rad})}{\pi D_3 (T_3 - T_6)} \quad (38)$$

A parabolic collector to generate power is modeled using MATLAB software with respect to optical and thermal analysis. A large number of characteristics including atmospheric pressure, environmental temperature and wind velocity can be employed in the presented model. With regard to required power and other restrictions, the absorber tube and the concentrator dimensions are accomplished.

4.2. Concentrator

In most of the collectors, the edge angle in concentrators is between 70 and 110 °C. The angle can be optimized with respect to manufacturing requirements and mechanical resistance [38]. In the present study, the edge angle is considered 90 °C which implies the minimized reflected beams and negligible tracking errors. The incident angle is normally assumed 0.95 for the mentioned edge angle.

Concentrator radiation coefficient is a function of the surface coating. Silvered glass and metalized acrylic are employed as the coating which have emissivity coefficient over 0.9. Metalized acrylic is cheaper but has lower corrosion resistance [38].

4.3. Absorber tube

The absorber tube is often made up of steel and covered by glass coating. Carbon steel absorber is well-known but 321H, 316L and 304L stainless steels are used for high temperatures. Recently, 321H has been used much since it decreases the bending problems and is resistant to hydrogen corrosion.

4.4. Coating on the absorber

The absorber is covered by a coating to maximize the absorption coefficient and minimize the emissivity coefficient.

Black chrome, black nickel and cermet are the popular coatings. Cermets have high temperature resistance. Black nickel has higher absorption coefficient and is cheaper but must be used less than 300 °C.

Four kinds of coatings are compared in Fig. 4 and their absorption and emissivity coefficients are indicated in Table 2.

The emissivity coefficient of black chrome is assumed as 0.11 when it is less than 0.11. The emissivity coefficient of Luz-Cermet is 0.05 in a case that it is less than 0.05. Figs. 4–8 are gained using MATLAB software.

Table 1
Constants for Eq. (36).

Re_D	C	m
1–40	0.75	0.4
40–1000	0.51	0.5
1000–200,000	0.26	0.6
200,000–1,000,000	0.076	0.7

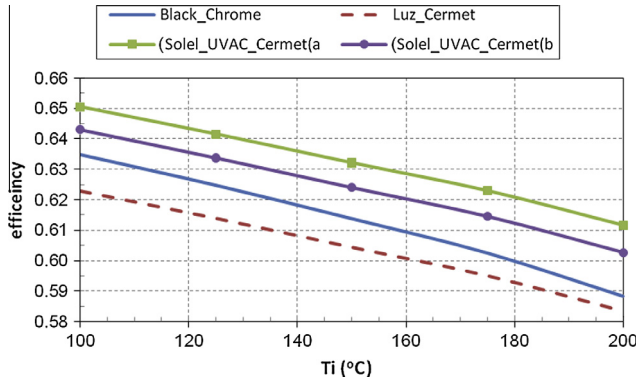


Fig. 4. Efficiency as a function of fluid temperature and different coatings.

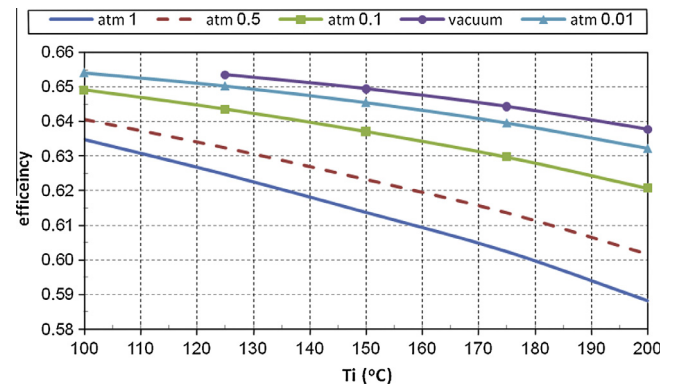


Fig. 5. Efficiency as a function of fluid temperature and annular space pressure.

4.5. Annular space pressure (between the absorber tube and the glass coating)

Fig. 5 represents the collector efficiency as a function of pressure in the annular space between the absorber tube and the glass coating. To reduce the thermal loss, vacuum state in the annular space is a solution and heat transfer occurs only by radiation.

As it is clear, the highest efficiency is the case of vacuum state in the annular space because there is not any convection heat transfer. The efficiency decreases with an increase in pressure. Because with increasing pressure, the heat transfer coefficient and conduction coefficient and therefore heat loss, will be increased. Also, more pressure has more effect on the efficiency (the slope is greater).

4.6. Gas in the annular space

The collector efficiency for air, hydrogen, argon and vacuum state in the annular space is depicted in Fig. 6. The vacuum state has the highest efficiency. Inert gases are in the next grade. However, is not much different with air.

Filling the annular space with inert gas improves stability and reduces hydrogen leakage from the working fluid [33]. As shown in Fig. 6 slopes, respectively, for the case of vacuum mode, argon that is an inert gas, air and hydrogen increased, means their effect on efficiency, respectively, will be increased. Hydrogen has noticeable effect on the efficiency.

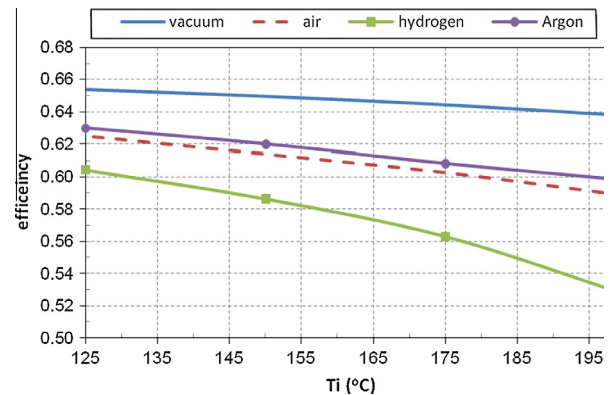


Fig. 6. Efficiency as a function of fluid temperature and the type of the gas.

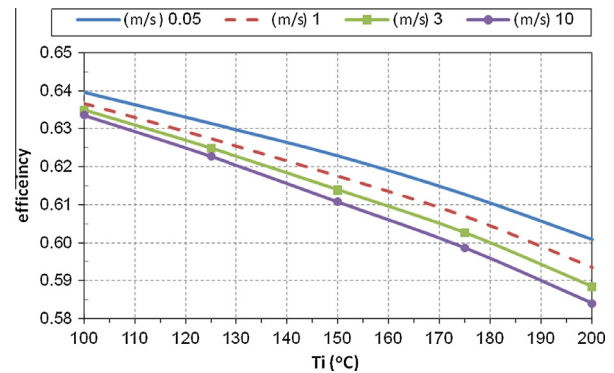


Fig. 7. Efficiency as a function of fluid temperature and wind velocity.

4.7. Glass coating

The glass coating reduces the thermal loss from the absorber. Low interval between the glass coating and the absorber causes high conduction thermal loss and high interval causes high convec-

tion thermal loss. The optimal interval is a function of Rayleigh number but it is often considered 8 mm.

Table 2
Coatings absorption and emissivity coefficients.

Coating	Absorption coefficient	Emissivity coefficient
Black_Chrome	0.94	$\varepsilon_3 = 0.0005333 \times T_3 - 0.0856$
Luz_Cermet	0.92	$\varepsilon_3 = 0.000327 \times T_3 - 0.065971$
Solel_UVAC_Cermet (a)	0.96	$\varepsilon_3 = 2.249 \times 10^{-7} \times T_3^2 + 1.039 \times 10^{-4} \times T_3 + 5.599 \times 10^{-2}$
Solel_UVAC_Cermet (b)	0.95	$\varepsilon_3 = 1.565 \times 10^{-7} \times T_3^2 + 1.376 \times 10^{-4} \times T_3 + 6.966 \times 10^{-2}$

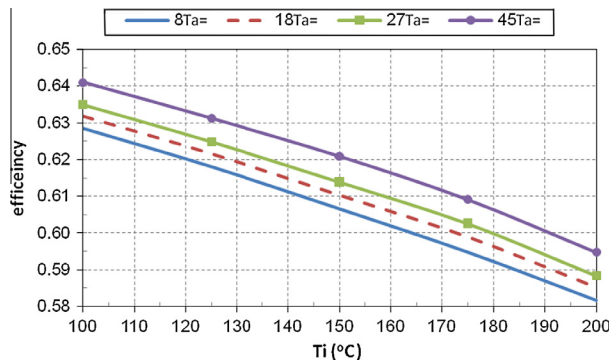


Fig. 8. Efficiency as a function of fluid temperature and environment temperature.

Table 3

Design parameters and operating conditions used for the case study.

Direct normal incident solar irradiation (I_b) (W/m ²)	800
Name of fluid	Therminol VP1
Ambient pressure (atm)	1
Wind speed (m/s)	3
Ambient temperature (°C)	27
Reflective	0.85
Incident	0.95
Edge angle (°)	90

4.8. Environmental conditions

Fig. 7 illustrates the efficiency as a function of wind velocity.

It is obvious that the efficiency decreases with an increase in wind velocity. With increasing wind velocity, the effect of forced convection will be increased. Thus, it causes increasing heat loss. It is clear that there is no wind, there is only natural convection. Fig. 8 depicts the efficiency as a function of the environment temperature.

It is inferred from Fig. 8 that the efficiency of the collector increases with increasing of environmental temperature. This increasing causes reducing the temperature difference between air and glass cover. Also, affects the Nusselt number and consequently convection coefficient. So increasing environment temperature decreases convection and radiation heat transfer from the glass cover to the environment.

4.9. The variables and design factors

The absorbed solar flux is a function of time. The program receives the time data (date). The program and is run for Tehran at 13:00 on July 2nd. The normal radiation is assumed at 800 W/m². The environment temperature, pressure and wind velocity are considered at 27 °C, 1 atm and 3 m/s, respectively.

The working fluid inlet temperature is assumed at 100 °C and Terminol vp1 is used as the working fluid.

Stainless steel grade 321H is applied for the absorber tube and the outer diameter of the absorber tube is considered as 4 cm. Black chrome is employed as coating on the absorber tube. The annular space between the absorber and the glass coating is assumed 8 cm and filled with air at 1 atm.

The reflective, incident and edge angles for the concentrator are assumed at 0.85°, 0.95° and 90°, respectively (see Table 3).

Table 4 shows collector's dimensions for different flows.

In the present project, a parabolic collector is modeled using MATLAB software. Wind velocity, air temperature and other effec-

Table 4

Collector's dimensions for different flows.

Flow rate (kg/s)	0.02	0.03	0.04	0.05	0.06	0.07	0.1
Collector length (m)	3.1	3.1	2.9	3	3.6	3.1	3
Collector aperture (m)	2.5	2.4	2.5	2.4	2	2.3	2.3
Collector power (W)	3502	3499	3495	3499	3498	3501	3504
Efficiency	0.564	0.587	0.602	0.607	0.607	0.607	0.634

tive parameters are discussed. The model is run for the maximum power at 3500 W.

5. Conclusion

Most collectors are designed to be coupled to a Stirling engine, are dish collectors. Also, they have high power or very low power and only one type of Stirling engine can couple to them. In this work, a collector is designed to connect to a Stirling engine with specified power. We obtain results such as collector length and aperture and other parameters for specified flow rate and conditions. Also, we obtain results of changing some conditions and parameters. Our collector has advantages. It is a parabolic-type and is designed for Stirling engine, in small scale and for domestic consumption with 3500 W of power. We can couple different type of Stirling engine to this collector. This system can use in other fields like air conditioning and solar water heater, in the following of generate electricity from Stirling engine. Most important thing, our parabolic collector has good and reasonable efficiency about sixty percent. This collector gives us many options to choose components and their material depending on financial issues and other things.

References

- [1] Kuznik F, Virgone J. Experimental assessment of a phase change material for wall building use. *Appl Energy* 2009;86(10):2038–46.
- [2] Garnier C, Currie J, Muneer T. Integrated collector storage solar water heater: temperature stratification. *Appl Energy* 2009;86(9):1465–9.
- [3] Sutthivirode K, Namprakai P, Roonprasang N. A new version of a solar water heating system coupled with a solar water pump. *Appl Energy* 2009;86(9):1423–30.
- [4] Anyanwu EE. Review of solid adsorption solar refrigeration II: An overview of the principles and theory. *Energy Convers Manage* 2004;45:1279–95.
- [5] Cheng ZD, He YL, Xiao J, Tao YB, Xu RJ. Three dimensional numerical study of heat transfer characteristics in the receiver tube of parabolic trough solar collector. *Int Commun Heat Mass Transfer* 2010;37(7):782–7.
- [6] Ceylan, Ergun A. Thermodynamic analysis of a new design of temperature controlled parabolic trough collector. *Energy Convers Manage* 2013;74:505–10.
- [7] Price H, Lufpert E, Kearney D, Zarza E, Cohen G, Gee, et al. Advances in parabolic trough solar power technology. *J Sol Energy Eng* 2002;124(2):109–25.
- [8] Calise F, Dentice d'Accadia M, Vanoli L. Thermoeconomic optimization of solar heating and cooling systems. *Energy Convers Manage* 2011;52:1562–73.
- [9] Kalogirou SA. The potential of solar industrial process heat applications. *Appl Energy* 2003;76:337–61.
- [10] Padilla RV, Demirkaya G, Goswami Y, Stefanakos E, Rahman MM. Heat transfer analysis of parabolic trough solar receiver. *Appl Energy* 2011;88:5097–110.
- [11] He Y, Xiao J, Cheng Z, Tao Y. A MCRT and FVM coupled simulation method for energy conversion process in parabolic trough solar collector. *Renew Energy* 2011;36:976–85.
- [12] Al-Nimr MA, Alkam M. A modified tubeless solar collector partially filled with porous substrate. *Renew Energy* 1998;13(2):165–73.
- [13] Naeeni N, Yaghoubi M. Analysis of wind flow around a parabolic collector (1) fluid flow. *Renew Energy* 2007;32(11):1898–916.
- [14] Naeeni N, Yaghoubi M. Analysis of wind flow around a parabolic collector (2) heat transfer from receiver tube. *Renew Energy* 2007;32(8):1259–72.
- [15] Li M, Wang LL. Investigation of evacuated tube heated by solar trough concentrating system. *Energy Convers Manage* 2006;47:3591–601.
- [16] Abouzeid M. Use of a reluctance stepper motor for solar tracking based on a programmable logic array (PLA) controller. *Renew Energy* 2001;23:551–60.

- [17] Xie WT, Dai YJ, Wang RZ. Numerical and experimental analysis of a point focus solar collector using high concentration imaging PMMA Fresnel lens. *Energy Convers Manage* 2011;52:2417–26.
- [18] Valmiki MM, Li P, Heyer J, Morgan M, Albinali A, Alhamidi K, et al. A novel application of a Fresnel lens for a solar stove and solar heating. *Renew Energy* 2011;36:1614–20.
- [19] Zhai H, Dai YJ, Wu JY, Wang RZ, Zang LY. Experimental investigation and analysis on a concentrating solar collector using Fresnel lens. *Energy Convers Manage* 2010;51:48–55.
- [20] Patnode AM. Simulation and performance evaluation of parabolic trough solar power plants. Master thesis. University of Wisconsin–Madison: College of Engineering; 2006.
- [21] Edenburn MW. Performance analysis of a cylindrical parabolic focusing collector and comparison with experimental results. *Sol Energy* 1976;18(5):437–44.
- [22] Pope R, Schimmel W. An analysis of linear focused collectors for solar power. In: Eighth intersociety energy conversion engineering conference, Philadelphia, PA; 1973. p. 353–9.
- [23] Ratzel A, Hickox C, Gartling D. Techniques for reducing thermal conduction and natural convection heat losses in annular receiver geometries. *Trans ASME J Heat Trans* 1979;101(1):108–13.
- [24] Clark J. An analysis of the technical and economic performance of a parabolic trough concentrator for solar industrial process heat application. *Int J Heat Mass Trans* 1982;25(9):1427–38.
- [25] Dudley V, Kolb G, Sloan M, Kearney D. SEGS LS2 solar collector-test results. Report of Sandia National Laboratories, SAN94-1884; 1994.
- [26] Thomas A, Thomas S. Design data for the computation of thermal loss in the receiver of a parabolic trough concentrator. *Energy Convers Manage* 1994;35(7):555–68.
- [27] Forristall R. Heat transfer analysis and modeling of a parabolic trough solar receiver implemented in engineering equation solver. National Renewable Energy Laboratory (NREL); 2003.
- [28] Stuetzle T. Automatic control of the 30 MWe SEGS VI parabolic trough plant. Master thesis. University of Wisconsin–Madison: College of Engineering; 2002.
- [29] Garcia-Valladares O, Velazquez N. Numerical simulation of parabolic trough solar collector: improvement using counter flow concentric circular heat exchangers. *Int J Heat Mass Trans* 2009;52(3–4):597–609.
- [30] Cheng Z, He Y, Xiao J, Tao Y, Xu R. Three-dimensional numerical study of heat transfer characteristics in the receiver tube of parabolic trough solar collector. *Int Commun Heat Mass Trans* 2010;37(7):782–7.
- [31] Gong G, Huang X, Wang J, Hao M. An optimized model and test of the China's first high temperature parabolic trough solar receiver. *Sol Energy* 2010. <http://dx.doi.org/10.1016/j.solener.2010.08.003>.
- [32] Kalogirou S. Solar energy engineering processes and system. 1st ed. Elsevier/Academic Press; 2009.
- [33] Sukhatme SP, Nayak JK. Solar energy principles of thermal collection and storage. 3rd ed. MacGraw Hill; 2008. p. 82–83.
- [34] Kalogirou SA. A detailed thermal model of a parabolic trough collector receiver. *Energy* 2012;48:298–306.
- [35] Kumaresan G, Sridhar R, Velraj R. Performance studies of a solar parabolic trough collector with a thermal energy storage system. *Energy* 2012;47(1):395–402.
- [36] Incropera Frank P, DeWitt David P. Fundamentals of heat and mass transfer. 5th ed. John Wiley & Sons; 2002.
- [37] Kongtragool B, Wongwises S. Optimum absorber temperature of a once-reflecting full conical concentrator of a low temperature differential Stirling engine. *Renew Energy* 2005;30:1671–87.
- [38] Kalogirou SA. Solar thermal collectors and applications. *Prog Energy Combust Sci* 2004;30(3):231–95.
- [39] Duffie JA, Beckman WA. Solar engineering of thermal processes. 3rd ed. New York: J. Wiley & Sons; 2006.
- [40] Cooper PI. The absorption of solar radiation on solar stills. *Sol Energy* 1969;12(3):333–46.